THE DESIGN, MANUFACTURE, AND TEST
OF A LOW PRESSURE QUICK RESPONSE
HIGH TEMPERATURE PRESSURE GAGE

A Thesis

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by

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INTRODUCTION

Pressure changes in working fluids are of utmost importance in many research and test procedures. Pressure indicators have been in use for many years, but performance defects have restricted their value even for laboratory and test stand work. These restrictions are due to any one of many defects which affect the operating limitations of most pressure receivers. Most pressure gages are sensitive not only to pressure, but also temperatures, to mechanical strains due to mounting effects and to mechanical vibrations in the walls of the pressure chamber. Most receivers heavy enough to withstand high pressures, or even moderate pressures have large masses or high inertia linkages which reduce the response to high or moderate frequency pressure changes.

With the above operating limitations in mind, a design was attempted which would have low inertia, greatly reduced response to temperatures, and a high response to moderately high frequency pressure changes.

THEORY AND DESIGN

For the pressure pickup, it was decided that the diaphragm, if small enough, would give satisfactory response to frequency changes, and at the same time if thin enough, it would be sensitive to small pressure changes.

To measure the diaphragm deflections electrical strain gages were selected. The resistance type strain gages are secured by cement to the member of which the strain is to be measured. This cement restricts the temperature to which the member can be exposed. With this limitation of temperature, a means of keeping this temperature down, or of isolating the strain gages from the hot gases was searched for.

Two diaphragms connected by a strut was at first considered, but this increased the inertia of the receiver, produced a pumping action in the barrel, reduced its response for given thickness of diaphragm, and reduced the natural frequency. After these considerations, it was decided to replace the outer diaphragm by a narrow beam. This substantially reduced the inertia of the system.

On this beam, the strain gages were to be mounted. The

length of the strut between the diaphragm and the beam was to be long enough to keep the temperature of the strain gages below the temperature of failure of the bonding material.

For the materials from which to construct the gage, the low expansion nickel alloys were considered in order to reduce the variations in the gage due to changes in temperature. Invar, a 36% nickel steel was considered best because of its low coefficient of expansion for temperatures up to 350° F. The 50% nickel alloy has a constant coefficient for temperatures up to 750° F but its coefficient is five times as great as the 36% alloy. (See Figure 1 for curve of coefficients versus temperatures) Invar was selected also because of its reasonably high melting point combined with its ability to resist corrosion when exposed to hot gases.

To determine what order of deflection the diaphragm-strut-beam system would take for various gas pressures, the loads were construed as being divided between the diaphragm and the beam inversely as the deflection of each member when subjected to a unit load.

The equation for the deflection of a clamped edge diaphragm with a center load is as follows:

$$w = \frac{Pr^2}{8\pi D} \log \frac{r}{a} + \frac{P}{16\pi D}(a^2-r^2)$$

$$w_{\underset{r=0}{max}} = \frac{Pa^2}{16\pi D}$$

D is the "Flexure of stiffness" and equals $\frac{Eh^3}{12(1-\mu^2)}$

$$= \frac{21 \times 10^6 \times 10^{-6}}{12(1-.084)}$$

$$= 1.91$$

When $P = 1$ $\quad w = \frac{.8^2}{16\pi(1.91)} = .0066"$

The equation for the deflection of a beam built in at both ends and loaded at the center is as follows:

$$w = \frac{P\, l^3}{192\ EI} \qquad EI = \frac{bh^3E}{12}$$

$$= \frac{.25 \times 21}{12}$$

$$= .437$$

When $P = 1$ $\quad w = \frac{4.096}{192 \times .437} = .05"$

Therefore the load will be divided as .0066 to .05. Precisely, the beam will take $\frac{.0066}{.0566}$ or .1165 of the load and the diaphragm $\frac{.05}{.0566}$ or .8835 of the load. (Equations from Reference 1)

To determine what portion of the total load will act on the strut from the diaphragm, it is necessary to determine what load alone at the center will give the same deflection as unit loading over the entire area of the diaphragm. From Reference 1 the equation for the deflection at the center of a diaphragm with uniform load over the entire area is as follows:

$$w = \frac{q\, a^4}{64\ D} = \frac{1 \times .4096}{64 \times 1.91} = .00335"$$

Equating this deflection to the deflection of the diaphragm for a single center loading and solving for the loading:

$$.00335 = \frac{P\,a^2}{16\,D}$$

$$P = \frac{.00335}{.0066} = .5075 \text{ lbs.}$$

Checking the deflections of beam and diaphragm for a 20 pounds per square inch pressure on the diaphragm:

For the beam: $w = .05 \times 20 \times .5075 \times .1165 = .0592''$

For the diaphragm: $w = 20 \times .00335 \times .8835 = .0592''$

Two strain gages were mounted on opposite sides of the beam in order to reduce temperature effects on the strain gages. The gages were bonded to the beam with a phenolic type thermo setting plastic cement capable of withstanding approximately 400° F.

MANUFACTURE

The selection of Invar for the gage material did not consider the machining difficulties of this tough and ductile metal. However, it is felt that the desirable physical and mechanical properties far outweigh its machining difficulties.

The barrel, and the diaphragm and beam retaining rings were machined from solid round stock three inches in diameter. Machining instructions from Reference (2) were followed closely and although the process was slow, no serious difficulties other than dulling of the cutting tools was incountered.

The diaphragm and beam were cut from .010 inch sheet Invar. The AB-11 type SR-4 strain gages were cemented to the beam, on opposite sides, one over the other, clamped and baked according to the baking schedule furnished with the cement. A good bond was made without any pockets under the gages.

After the gage was assembled, the strut was peened to the diaphragm and beam. See Figures 3 and 4 for pictures of gage assembled and disassembled.

DETERMINATION OF THE NATURAL FREQUENCY

Considerable time was spent in trying to develop an equation for the theoretical determination of the natural or resonant frequency of the diaphragm-strut-beam system. It was decided that the system was rather complex and since accurate determination was not considered necessary to the accurate calibration of the gage. With this in mind an experimental determination of the natural frequency was obtained as follows:

The strain gages were connected to the Strain Recorder and the Recording Oscillograph. The pressure gage was tightly clamped in a large bench vise. The system was made to vibrate at its natural frequency by dropping a small drill rod on the diaphragm and the oscillations recorded on the recording oscillograph. Several runs were made with good duplication of results. The natural frequency, thus determined, was 527 cycles per second.

Using the equation for a clamped edge diaphragm, and assuming the strut and beam effect to be that of a constant in this equation, it was possible to predict within reasonable accuracy the natural frequency of this system with diaphragms of different thicknesses.

$$f = K \frac{1}{2} \sqrt{\frac{Eg}{\rho}} \quad \frac{h}{a^2}$$

For a plot of frequencies versus diaphragm thickness see Figure 2.

STATIC TESTING

To test the gage under static conditions, it was mounted on the end of a cylinder made from a short piece of two inch pipe. It was held in place by means of its retaining collar and two holddown bolts. (See Figure 6 for test equipment set-up)

The air pressure in the cylinder was controlled by means of a needle valve and the pressure was measured by means of a mercury manometer sixty inches high.

For temperature calibration, an iron-constantan thermocouple was mounted on the barrel of the gage and a hot plate used to slowly and uniformly heat the gage to different temperatures.

Several sets of readingswere taken for each temperature and results were closely duplicated in the room temperature runs, but not so closely duplicated in the higher temperature runs.

Total strain versus pressure for each temperature is shown in Figure 7. Strain per inch of mercury pressure for various pressures and temperatures is shown in Figure 8. Figure 7 was constructed from the averaged curves of Figure 8.

DYNAMIC TESTING

For dynamic testing the pressure gage was mounted in a T fitting in the exhaust stack of a six cylinder Chevrolet engine. See Figure 5 for schematic drawing of test set-up. The gage was forty-one inches downstream from the exhaust manifold. In this connection, it might be mentioned that a closer mounting of the gage to the exhaust manifold would show much sharper pressure changes. The length of exhaust stack leading to the gage has some damping effect on the individual cylinder exhaust pressures.

The engine was equipped to operate on three cylinders through the use of a devise for preventing the valves of three cylinders from opening.

The engine was driven by the dynamometer for the first run. Runs 2, 3, and 4 were at increasing speeds with three cylinders and with a light load. These runs showed an increase in pressure variations from 1.04 inches of mercury at 265 rpm to 3.56 inches of mercury at 629 rpm. The first run showed a pressure variation of only 1.15 inches of mercury at 700 rpm.

Runs 5 through 15 were made at approximately the same

rpm but with generally increasing loads. These runs showed a general increase in pressure variation with increasing load.

The last two runs were made at high rpm (1010 and 1435) and they showed a decrease in pressure with increasing rpm and same load.

Very few absolute pressures were obtained and these were believed to be in error due to the rapid creep of the "zero" of the gage with changing temperatures.

Table I shows the data for all runs as reduced from traces of the recording oscillograph.

CONCLUSIONS

Static test results were quite satisfactory. The gage followed, with reasonable accuracy, an almost linear variation of pressure versus strain for all test temperatures and static pressures. Errors were less than 7.5% for pressures up to 44 inches of mercury.

For the dynamic tests, "zeroing" the gage was all but impossible. It was necessary to stop the engine to zero the gage after the desired test temperature was reached. During this short time interval the gage cooled slightly, or as in some cases, continued to increase in temperature due to the hot exhaust stack. When the engine was again running at the proper rpm and load, the gage would change temperature again and give wide variations in readings. During any one run the creep of the zero on the oscillograph trace could be seen to move considerably as the gage temperature changed even though the exhaust temperature would remain reasonably constant. Run 23 showed a variation over the length of a four second trace of nine tenths of an inch of mercury per second.

The difference in pressure (high and low) over very

short periods or between adjacent cycles is shown quite accurately on the traces of the recording oscillograph. The change of strain per inch of mercury pressure is reasonably constant and is plotted for test temperatures in Figure 8.

Another method of measurement of the diaphragm deflection occurred to the author and seems worthy of mention for further research. This consists of a very small (very low inertia) linear variable differential transformer secured on the end of the strut to replace the beam. Motion of the strut would move the very light core or casing of the transformer and potentiometer readings would show the linear displacement of the diaphragm center.

RUN #	RPM	NUMBER CYLINDERS	CYCLES PER SECOND	LOAD CONDITION	PRESSURE VARIATION	EXHAUST TEMP. °F	GAGE TEMP. °F	AVERAGE ? GAGE PRES.
1	700	3	17.5	DRIVEN	1.15"Hg	100	96	.46"Hg
2	265	3	6.6	LIGHT	1.04	183	97	
3	400	3	10	INCREASED	2.89	330	108	
4	629	3	15.7	INCREASED	3.56	400	111	
5	440	6	22	NO LOAD	1.0	330	121	
6	435		21.7	LIGHT	1.6	365	131	
7	520		26	LIGHT	1.9	490	150	
8	530		26.5	INCREASING	1.85	505	170	4.55
9	545		27.2		2.28	580	182	
10	549		27.5		2.5	610	191	
11	550		27.5		2.59	635	203	
12	550		27.5		2.48	645	214	
13	545		27.3		2.62	653	226	
14	540		27		2.62	657	238	
15	546		27.3	SAME	2.64	660	250	
16	550		27.5	SAME	2.64	666	260	
17	708		35.4	MEDIUM	4.88	705	272	
18	710		35.5	SAME	4.88	715	280	5.87
19	716		35.8	SAME	4.40	734	301	
20	710		35.5	INCREASED	5.00	705	304	
21	1010		50.5	LIGHT	3.6	730	305	
22	1435		70.7	LIGHT	1.54	740	304	
23	250		12.5	MEDIUM	5.65	480	289	
24	190		9.5	MEDIUM	4.12	450	282	2.86

TABLE I

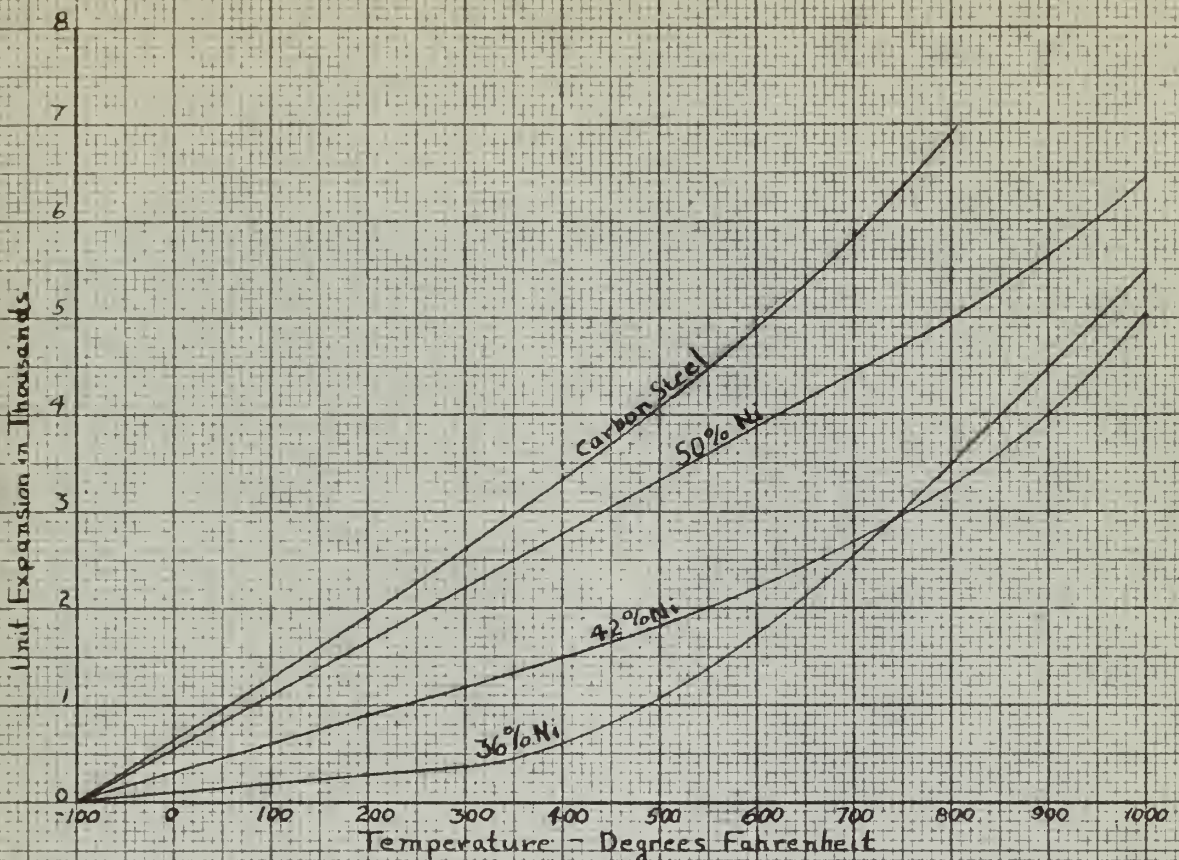


FIG. 1 Expansion curves for some iron-nickel alloys

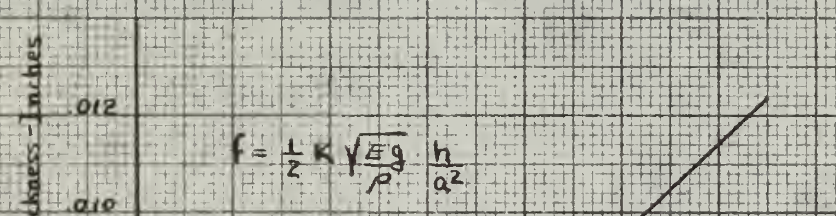


FIG. 2 Frequency vs Diaphragm Thickness

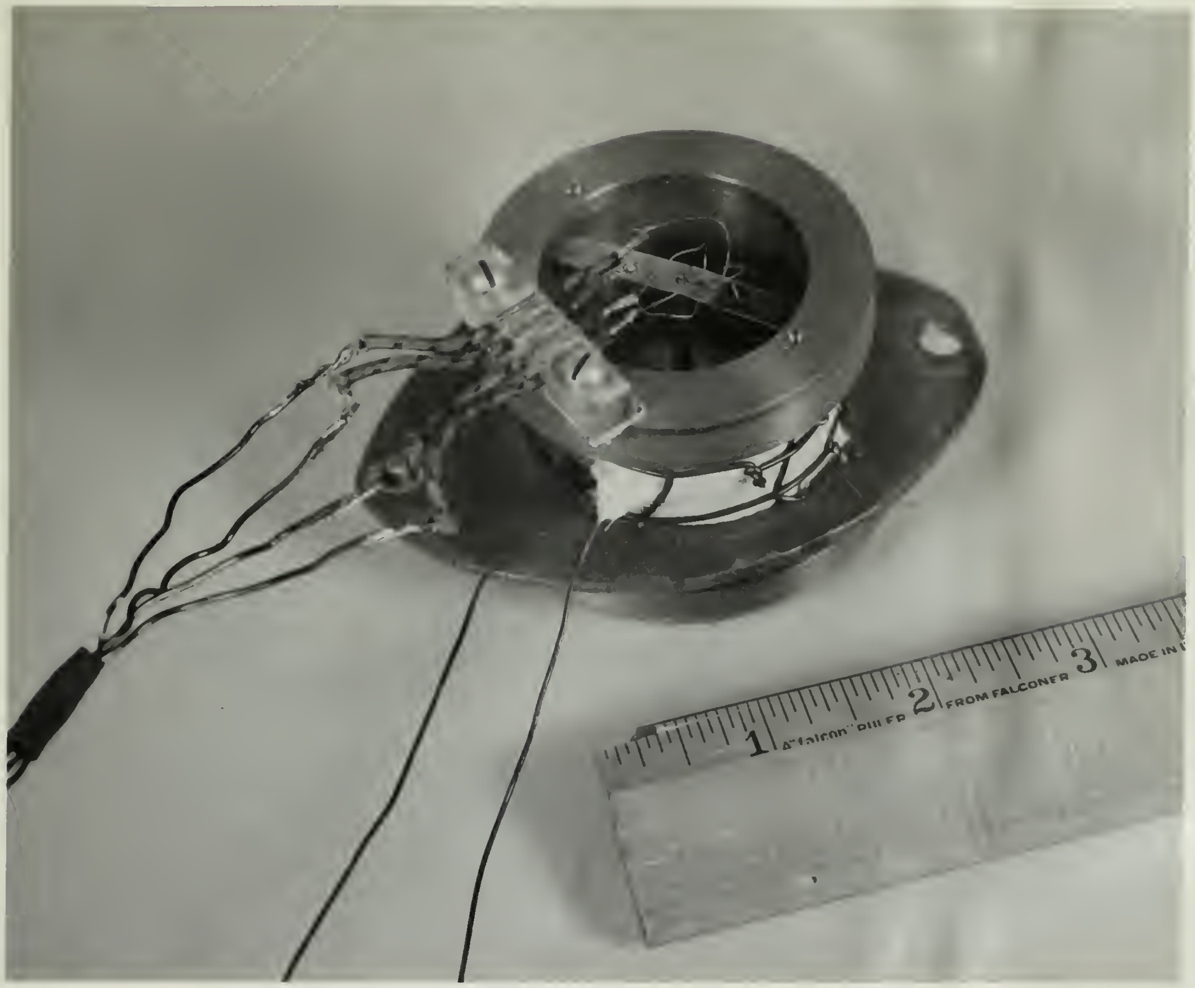
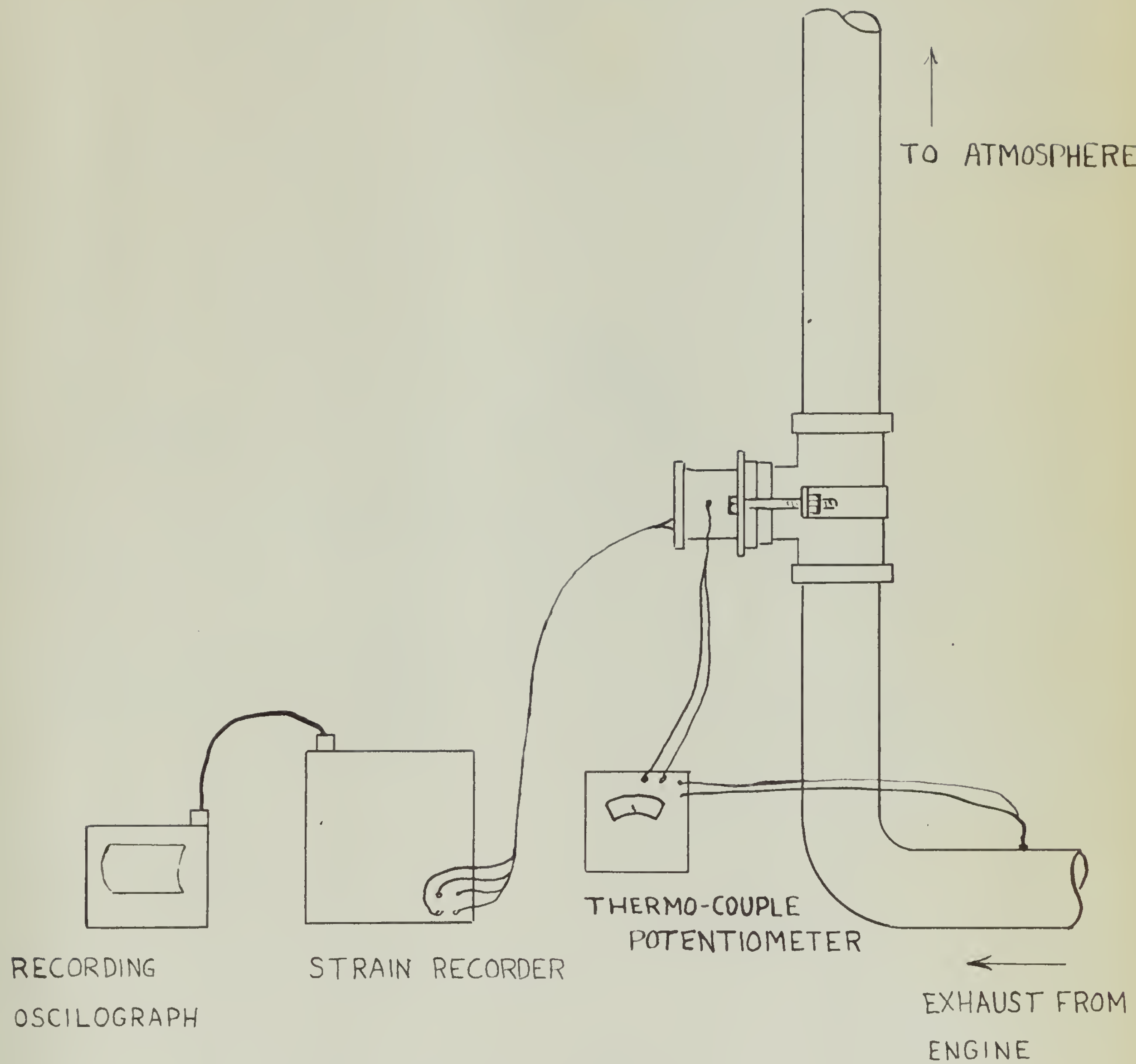


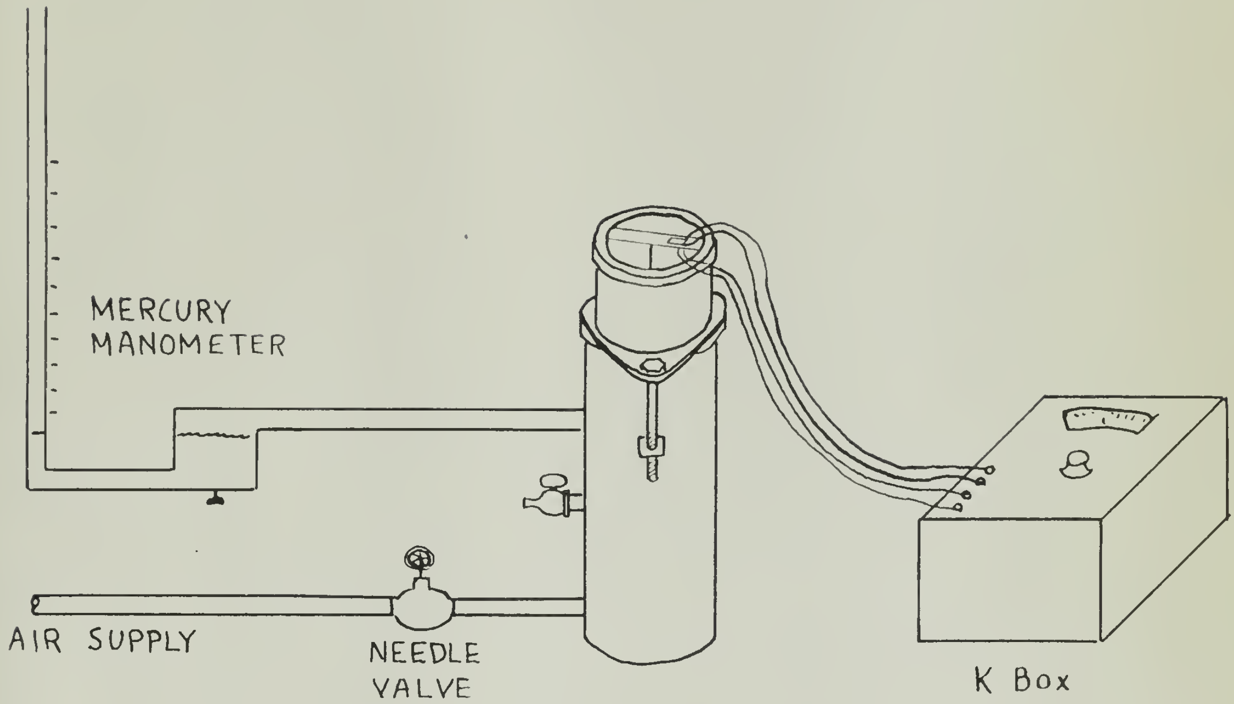
Figure 3   Pressure Gage Assembled

Figure 4 Pressure Gage Disassembled



SET-UP FOR DYNAMIC TEST

FIG. 5



SET-UP FOR STATIC TEST

FIG. 6

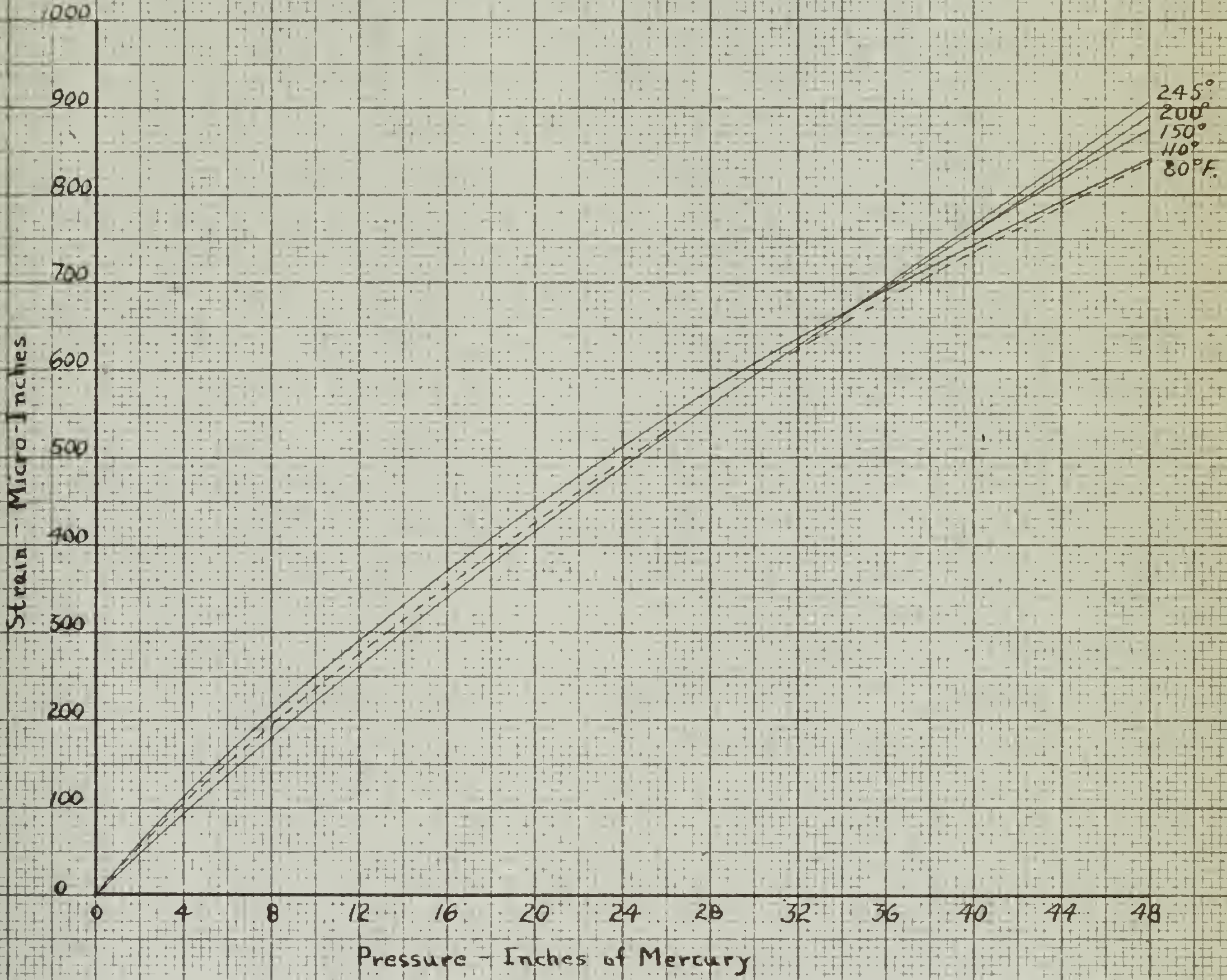


FIG. 7 Total Strain vs Pressure

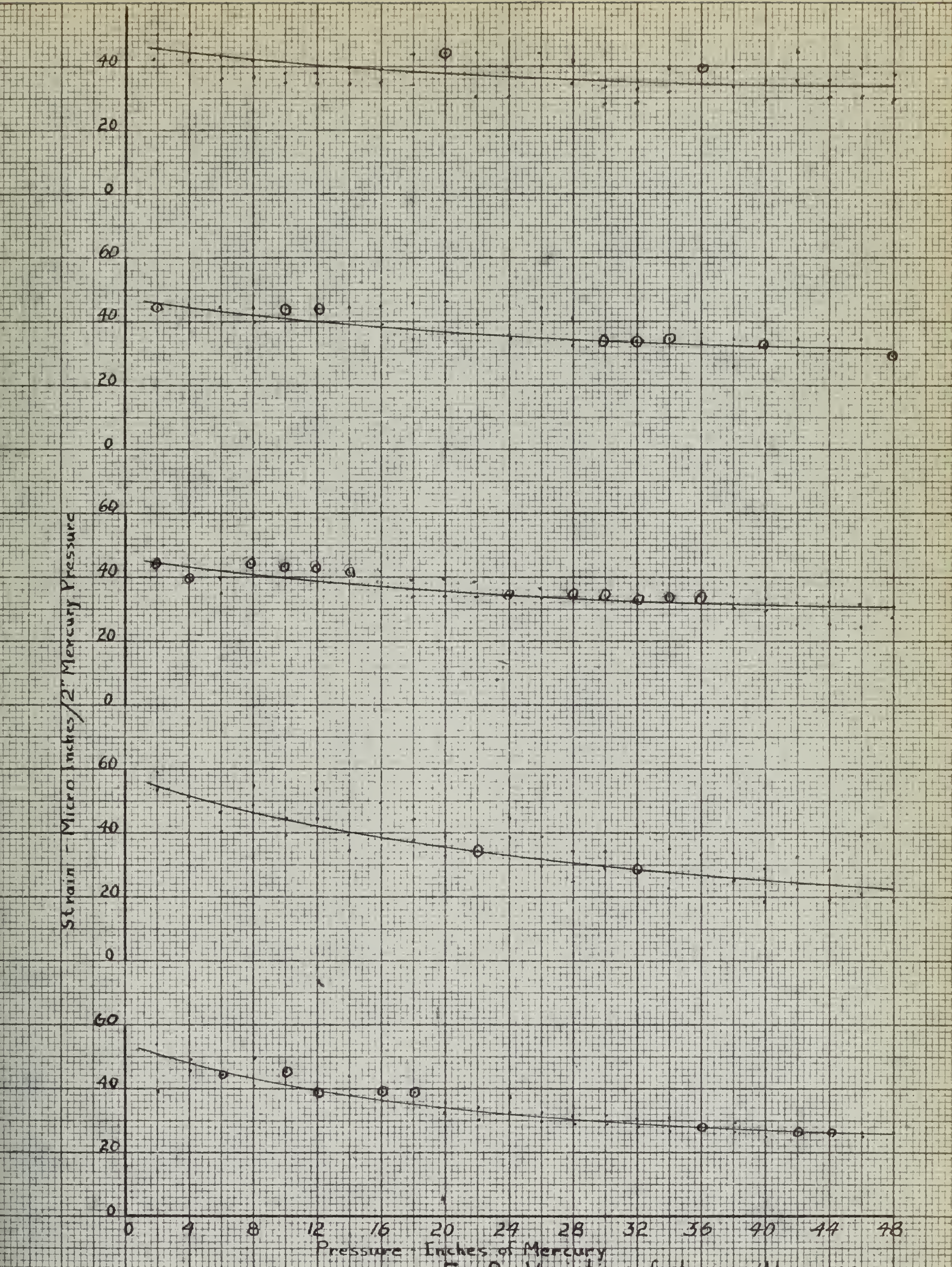


FIG. 8 Variation of strain with pressure

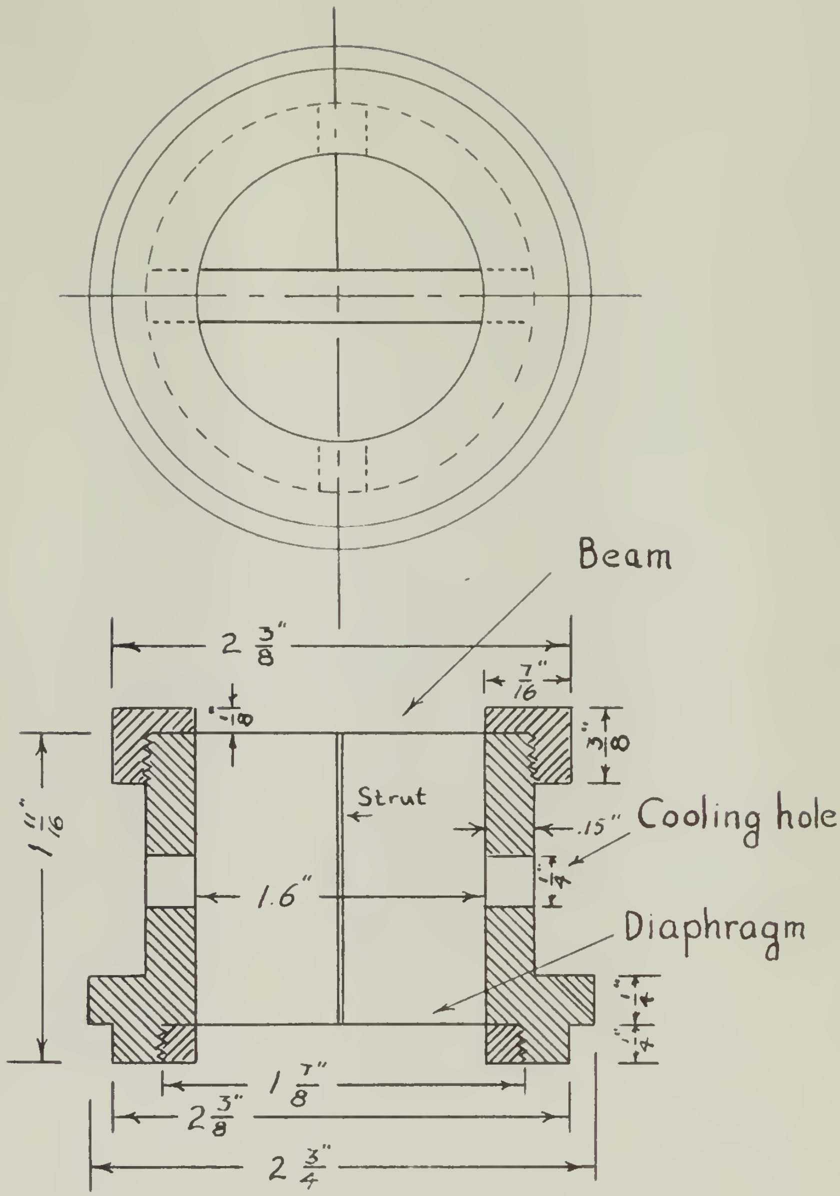


FIG. 9 Scale drawing of gage

APPENDIX

NOTATIONS AND DIMENSIONS

E	Modulus of Elasticity, lbs/sq.in.
h	thickness, in.
μ	Poisson's Ratio
b	Width of beam, in.
q	Load/ unit area, lbs/sq.in.
P	Concentrated load, lbs.
w	Deflection, in.
ρ	Density, lbs/cu.ft.
g	Acceleration due to gravity
a	Radius, in.
r	Distance from center along radius, in.
I	Moment of inertia, $in.^4$

FOR THE GAGE TESTED:

h = .01 inches

b = .25 inches

μ = .290

a = .8 inches

ρ = 508 lbs/cu.ft.

$I = .0208 \times 10^{-6}$ $in.^4$

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The design, manufacture, and test of a low pressure quick response high temperature pressure gage.